Rod Millen

Special Vehicles

JOINT TACTICAL ELECTRIC VEHICLE DIFFERENTIAL DEVELOPMENT

ACTIVE DIFFERENTIAL

FINAL REPORT

CALSTART PROPULSION SYSTEM DEVELOPMENT PROJECT FOR ADVANCED HYBRID RECONNAISSANCE VEHICLES

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AQ

"Joint Tactical Electric Vehicle Differential Development, active differential" final Report

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JOINT TACTICAL ELECTRIC VEHICLE DIFFEREENTIAL DEVELOPMENT

ACTIVE DIFFERENTIAL FINAL REPORT

CALSTART PROPULSION SYSTEM DEVELOPMENT PROJECT FOR ADVANCED HYBRID RECONNAISSANCE VEHICLES

INTRODUCTION

As part of Rod Millen Special Vehicle's contract number MDA972-95-1-0011 (ARPA RA94-24 Program), effective April 1, 1996, under the aegis of CALSTART's *Propulsion System Development Project for Advanced Hybrid Reconnaissance Vehicles* effort, the Joint Tactical Electric Vehicle (JTEV) differentials were to be evaluated with the intent of improving the overall performance level of the vehicle.

BACKGROUND

JTEV's current drive system incorporates a Weismann locker differential system to allow the rotational rate of the wheels to vary from left to right across the vehicle, and allows the wheel with the most traction to receive the full driving torque available. This type of drive was selected for maximum mobility in an off-the-shelf system. This system has the advantage of always providing drive to the ground, but under very poor or rapidly changing traction conditions, it has the potential to overload the driving wheel and break traction, thus shifting all drive to the opposite wheel. This could cause instability due to the vehicle's tractive effort center rapidly alternating from left to right. In order to further demonstrate the tactical utility of hybrid electric vehicle technology, it was deemed desirable to attempt to improve the differential system with the intent of achieving a system capable of producing the absolute maximum tractive effort in all possible conditions, without adversely affecting JTEV's handling characteristics.

DESIGN SUMMARY

This project necessitated the design of a system capable of transmitting the full torque available at JTEV's differential through one wheel, and also capable of behaving as a completely open differential, inducing no torque bias across the axle. In addition, this system had to be able to fit in the space envelope available on JTEV, and be actuated by less than 1500 psi, so as to allow use of commonly available hydraulic components and seals. The torque transmission across the differential clutch pack was calculated to be approximately 225 ft-lbf, and the pressure to release the clutch pack was calculated to be approximately 750 psi. These calculations are detailed in Appendix A.

This project culminated in the design of a hydraulically actuated variable load clutch system capable of being fitted to JTEV. The unit consists of a set of friction discs and a hydraulic cylinder integrated in a very compact package. This clutch system is designed to function with a conventional, spider-gear type differential, and is intended to vary the torque transmission across the wheels on a given axle. The whole differential system is then to be integrated with a computer control through an hydraulic actuator. The control of differential slip allowed by this type of a system can give the vehicle maximum tractive effort (both in acceleration or climbing and also under braking) without introducing the poor cornering behavior associated with locked differentials. In the event of a loss of hydraulic pressure, this unit is designed to lock completely. This feature ensures that the vehicle maintains it's full drive capability. This situation would adversely impact the vehicles' cornering performance, but being able to pull out of the bog is more important than having optimum handling in tight corners. An assembly drawing and layout schematic is included as Appendix B.

Improving vehicle stability and traction using a computer controlled differential can be approached in a number of ways. The two most obvious are using a closed loop control system which simply attempts to equalize the torque transmitted by the left and right wheels of the vehicle, and an open loop control system which monitors operational parameters of the vehicle and regulates differential lockup with the goal of not impairing vehicle handling. Although a closed loop system seems a more elegant solution, such an approach is fraught with difficulties, including the difficulty of accurately and reliably monitoring axle torque, and the very real possibility of control system instability. For these reasons, an open loop control system would be the primary avenue of development for further work on this project.

TESTING

Bench testing of this unit was conducted to determine its ability to transmit torque, and it's ability to be pre-set to slip at different torque levels. The friction plates were lubricated with Mobil 1 15w-50 synthetic oil prior to assembly. The unit was assembled with varying shim stacks in order to vary the spring preload on the friction discs. Each assembly was then tested to determine its torque transmission ability. Torque transmission varied between 10 and 450 ft-lbf. Typically, static friction was five to ten percent higher than dynamic friction, but the dynamic friction measurements were not at all precise, especially at higher torque values.

Test results are summarized in Appendix C.

CONCLUSIONS

Bench testing shows that this unit as currently designed has the capability to function as the heart of an active differential system on JTEV. Further analysis is warranted prior to on-vehicle testing.

RECOMMENDATIONS

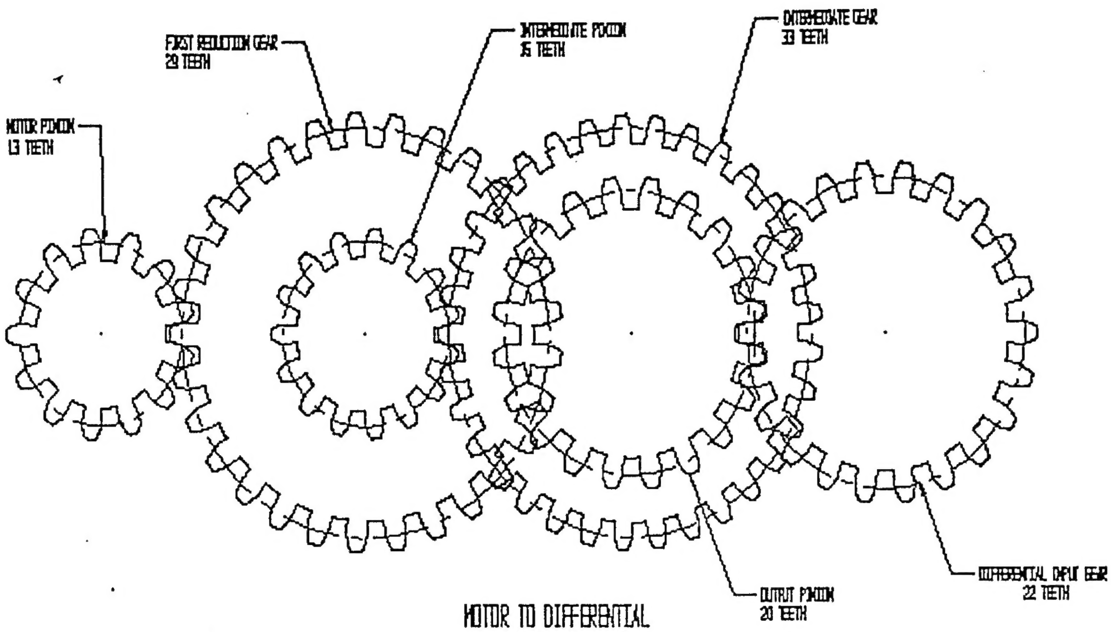
This system has the potential to improve the performance of the Joint Tactical Electric Vehicle under traction-limited conditions. Further analysis will be required to determine if the unit will be able to handle the thermal loading that might be imposed in the most severe situations. A number of options exist if the thermal issues prohibit use of this exact design. Among them are forced oil cooling through the clutch pack housing, and use of a heat exchanger in the hydraulic system.

Vehicle testing is required to validate the functionality of this system. The first phase of such testing should be to select and implement a suitable spider-gear differential, and then install the system on JTEV with only a rudimentary control system. This system should be able to produce no slip, 100% slip and 50% slip across the differential. During vehicle testing the system should be instrumented to evaluate thermal loading, torque split, and overall tractive effort.

APPENDIX A

Active diff torque and pressure calculations

JTEV Torque



MOTOR TO DIFFERENTIAL
GEAR REDUCTION

$T_{motor} := 108 \cdot newton \cdot m$

$R_1 := \frac{29}{13} \quad R_2 := \frac{33}{15} \quad R_3 := \frac{22}{20}$

$T_{diff} := T_{motor} \cdot R_1 \cdot R_2 \cdot R_3$

$T_{diff} = 430.024 \cdot ft \cdot lbf$

Diff clutch torque req'd = 450 ft-lbf, based on JTEV

Clutch pack sizing:

$\mu := .1$

$M := \frac{450}{2} \cdot ft \cdot lbf$

$D := 4.0 \cdot in$

$d := 3.5 \cdot in$

torque transmitted through clutch pack is half of axle torque

$$M = \mu \cdot L \cdot \frac{D^3 - d^3}{D^2 - d^2}$$

$$L := \frac{-M}{\left[\frac{-\mu}{\left(D^2 - d^2\right)} \cdot D^3 + \frac{\mu}{\left(D^2 - d^2\right)} \cdot d^3\right]}$$

L=axial load required on clutch pack

$L = 4.793 \cdot 10^3 \cdot lbf$

$D_{cyl} := 3.0 \cdot in$

$t := .25 \cdot in$

$d_{cyl} := .9 \cdot in$

$$A_{cyl} := \frac{{D_{cyl}}^2 - {d_{cyl}}^2}{4} \cdot \pi$$

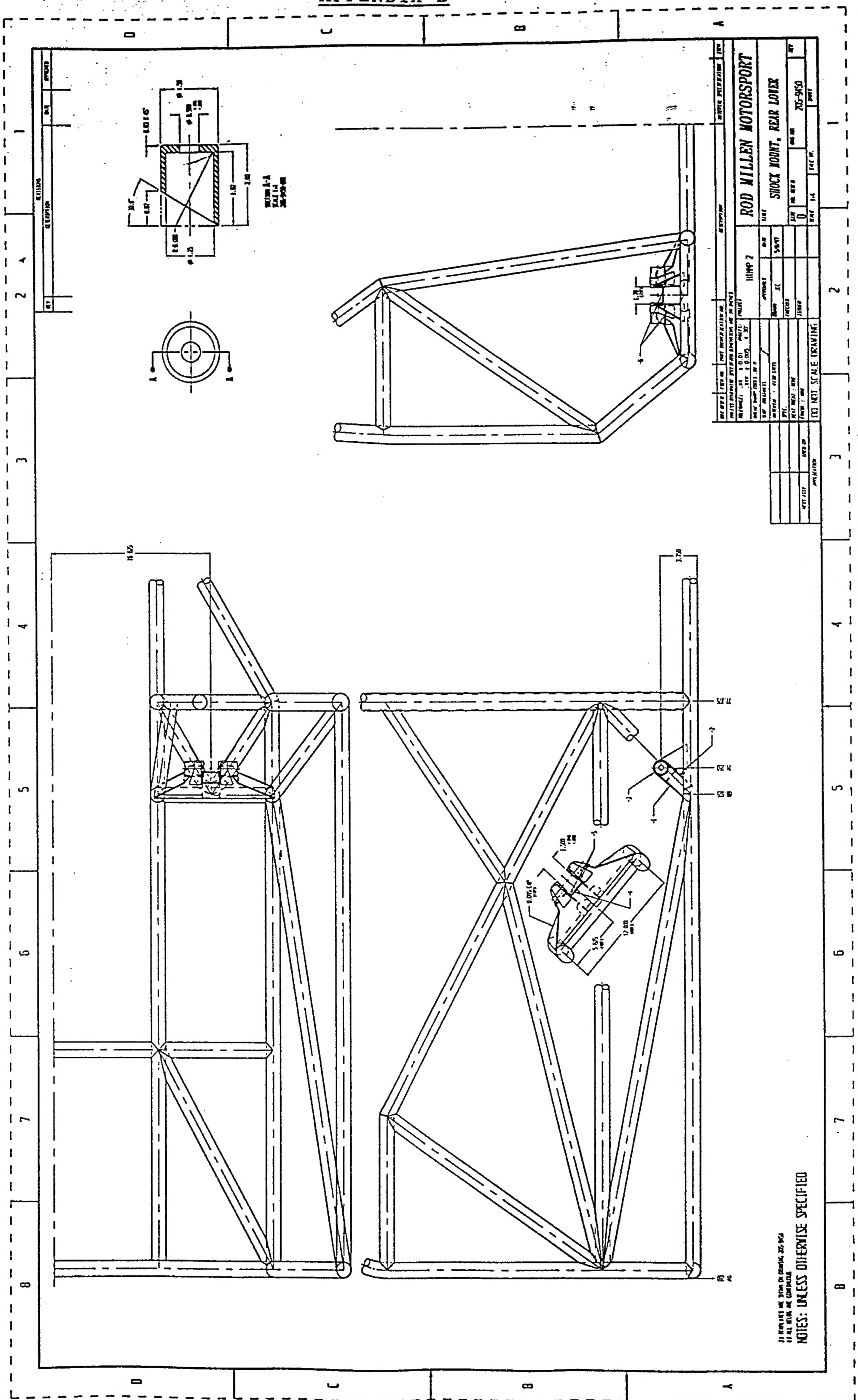
$$P := \frac{L}{A_{cyl}}$$

$P = 745.117 \cdot psi$

$$\sigma := \frac{P \cdot D_{cyl}}{2 \cdot t}$$

$\sigma = 4.471 \cdot 10^3 \cdot psi$

APPENDIX B


ROD MILLEN MOTORSPORT
SHOCK MOUNT, REAR LOWER
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NOTES: UNLESS OTHERWISE SPECIFIED

APPENDIX C

Bench test results

Temperature:	70 deg F
Component condition:	New
Lubrication:	Mobil 1 15W50 synthetic oil, liberally applied to friction and floater plates
Instrumentation:	Western Auto p/n H2837 beam-type torque wrench and Snap-on YA290 torque multiplier

Test results:

run #	shim thickness	static torque	dynamic torque
1	.115 inch	10 ft-lbf	10 ft-lbf
2	.104	160 ft-lbf	160 ft-lbf
3	.099	300 ft-lbf	280 ft-lbf
3	.092	400 ft-lbf	370 ft-lbf
3	.084	430 ft-lbf	415 ft-lbf
3	.077	500 ft-lbf	470 ft-lbf

note that torque measurements are +/- 10% due to difficulty in measuring